CONTINUOUSLY VARIABLE-TRANSMISSION

The present invention relates to a continuously variable transmission in accordance with the preamble of Claim 1. A
5 transmission of this type is generally known and is used to transmit mechanical power between the primary and secondary pulleys of the transmission, it being possible for a transmission ratio of the transmission with which a couple or rotational speed is transmitted to be varied continuously
10 within a certain range. As is known, the drive belt is clamped between the two pulley discs, which are substantially frustoconical or conical in form, of the respective pulleys; in the present invention, the transmission ratio of the transmission is defined as the ratio between an effective
15 radial position of the drive belt on the secondary pulley and an effective radial position of the drive belt on the primary pulley, which positions are respectively also known as the secondary running radius and the primary running radius. To enable these running radii, and therefore the transmission
20 ratio, to be varied, at least one of the pulley discs of each pulley is arranged such that it can be moved in the axial direction.
By way of example, it is known from European patent application EP-A-1 218 654 that the two axially oriented
25 forces with which the drive belt is clamped between the pulley discs of the pulleys, referred to below as the primary clamping force and the secondary clamping force, respectively, represent a crucial factor in determining the torque which can be transmitted between the pulleys via
30 frictional forces between the pulleys and the drive belt, while the ratio between these clamping forces is a crucial factor in determining the transmission ratio. It should be noted that the minimum clamping force required for each pulley to transmit a torque which is supplied can be
35 approximated by the following equation:

$$Kp = \frac{Tp * \cos(\lambda)}{2 * \mu_T * Rp} \qquad (1)$$

In this equation, Kp is the minimum clamping force which is to be exerted on the drive belt by a pulley disc of the primary pulley in order for a primary torque Tp which is supplied to this primary pulley to be transmitted, i.e.
5 virtually without any slip between the drive belt and the respective pulley disc in the tangential or circumferential direction, with a tangent line on the pulley disc at the location of an effective contact point between this pulley disc and the drive belt forming a contact angle $\lambda$ with the
10 radial direction, the said point of contact being located at a radial distance Rp from a centre of rotation of the pulley, which distance corresponds to the said primary running radius, with an effective coefficient of friction $\mu_T$ between the drive belt and the pulley disc prevailing in this
15 tangential direction.

The minimum secondary clamping force Ks required can be calculated in a corresponding way from a secondary torque Ts and a secondary running radius Rs. However, since the ratio between the torque and the running radius Tp/Rp, and Ts/Rs,
20 respectively, ignoring possible losses, is inevitably equal for the two pulleys, the minimum secondary clamping force required is equal to the said minimum primary clamping force required.

In practice, a ratio between the primary and secondary
25 clamping forces, referred for short as the clamping force ratio, will, however, have to be significantly higher or lower than 1, in order to enable a defined desired transmission ratio to be realised. The clamping force ratio required for an equilibrium state of the transmission, i.e. a
30 constant transmission ratio, is referred to below as the equilibrium clamping force ratio, denoted here as the KpKs ratio. For the known transmission, the equilibrium clamping force ratio has a different value in different transmission ratios, this equilibrium clamping force ratio generally being
35 greater than 1 at least in numerically the lowest transmission ratio, i.e. Overdrive, and less than 1 at least in numerically the highest transmission ratio, i.e. Low. The relationship between the transmission ratios of the transmission and the associated equilibrium clamping force

ratio for a constant transmission ratio is referred to below
as the KpKs curve for short. In a non-equilibrium state of
the transmission, in which the transmission ratio is
decreasing or increasing, the clamping force ratio required
5  is raised or lowered, respectively, with respect to the said
equilibrium clamping force ratio, the extent to which a
clamping force ratio which is actually effected, referred to
here as the FpFs ratio, deviates from the equilibrium
clamping force ratio being the crucial factor in determining
10 the speed at which the transmission ratio changes.
In the equilibrium state of the transmission, therefore, the
lower out of the primary and secondary clamping forces has to
be at least equal to the minimum level required for the
torque transmission, while the higher of the clamping forces
15 is then given by the equilibrium clamping force ratio, i.e.
the KpKs ratio. If the KpKs ratio deviates from 1, therefore,
at least one of the clamping forces will adopt a higher level
than the said minimum required level in order to realise the
equilibrium state.
20 It should be noted that the clamping forces are effected with
the aid of suitable, generally known actuation means, which
usually act on the axially moveable disc of a pulley, such as
a hydraulically acting piston/cylinder assembly or an
electrically driven threaded spindle. The clamping forces for
25 each pulley act on the drive belt over the length of a part
of the drive belt which is clamped between the respective
pulley discs. In the definition according to the invention,
the said length is quantified per pulley as the angle which
is enclosed by the respectively clamped part of the drive
30 belt and is referred to as the primary and secondary belt
angle. In this case, the sum of the primary belt angle and
the secondary belt angle is, of course, equal to $2\pi$, i.e. the
arc of a circle described by the drive belt for each pulley
together always form a complete circle.
35 Moreover, in the known transmission, at least for one of the
pulleys, the said minimum required clamping force is
increased and/or multiplied by a safety factor which defines
the clamping force which is ultimately desired and actually
applied. The effect of an increase of this nature is to

ensure that any inaccuracy in the parameters from equation
(1) or, for example, an excessively rapid increase in the
torque supplied does not cause the abovementioned slipping of
the push belt and a pulley.
5   In a control system which is in widespread use in practice
and is generally known as the (VDT) master-slave control, the
starting point is a desired secondary clamping force $Ks_{DV}$,
which is given by multiplying the minimum secondary clamping
force Ks required, calculated for example using equation (1),
10  by a safety factor Sf:

$$Ks_{DV} = Sf*Ks \qquad (2)$$

The respective other clamping force, in this case the desired
15  primary clamping force $Kp_{DV}$, which is required in order to
realize a desired and constant transmission ratio then
follows by multiplying the desired secondary clamping force
$Ks_{DV}$ by the numerical value given by the KpKs curve at the
transmission ratio, i.e.:
20

$$Kp_{DV} = KpKs*Ks_{DV} \qquad (3)$$

For the transmission which is predominantly used in practice
with a contact angle of approximately 11 degrees, the
25  equilibrium clamping force ratio, when using a safety factor
of approximately 1.3, which is generally the minimum value
used, and depending on the transmission ratio, has been found
typically to vary between approximately 0.9 in Low and 1.8 in
Overdrive. A relationship of this nature between the
30  transmission ratios of a transmission and the associated
equilibrium clamping force or KpKs ratio is referred to as
the KpKs curve. Although other parameters, such as the torque
level, a rotational speed of the primary axle, the
temperature, etc. do influence the KpKs curve, according to
35  the invention these parameters can in a first approximation
be considered negligible compared to the influence of the
safety factor.
To change the transmission ratio, the primary clamping force
Kp which is actually effected is increased or reduced with

respect to its desired value $Kp_{DV}$ in order to cause the transmission ratio to change toward Overdrive (increase in Kp) or Low (reduction in Kp). In this context, the more the FpFs ratio deviates from the KpKs ratio, the more quickly the
5   transmission ratio will change.

It follows from equations (2) and (3) that in the Master-Slave control the safety factor should be sufficiently high for it to compensate not only for the above-mentioned possible inaccuracy in the parameters from equation (1) but
10  also for a numerical value of lower than 1 which is given by the KpKs curve in Low. This latter aspect can be explained by not using a safety factor in equation (2), i.e. by setting the factor Sf to be equal to 1. In that case, it follows from equations (2) and (3) that if said KpKs numerical value is
15  lower than 1, the desired primary clamping force $Kp_{DV}$ is lower than the minimum secondary clamping force Ks required. However, as has already been noted, the minimum primary clamping force Kp required to transmit the supplied torque corresponds to the minimum secondary clamping force Ks
20  required, so that the latter desired primary clamping force $Kp_{DV}$ is insufficient and the drive belt will slip with respect to the primary pulley. In other words, it follows from the requirement that the desired primary clamping force $Kp_{DV}$ must be at least equal to the minimum secondary clamping force Ks
25  required that if the KpKs ratio can adopt a value of lower than 1, the safety factor Sf in the Master-Slave control should be at least equal to 1 divided by the lowest KpKs ratio which occurs, $[KpKs]_{Low}$ for short.

In addition, the safety factor in the Master-Slave control
30  serves to realize a type of reserve in addition to the minimum required level of primary clamping force Kp, with the result that it becomes possible to change the transmission ratio toward Low by reducing the primary clamping force Kp without this parameter becoming lower than the minimum
35  primary clamping force Kp required to transmit the torque supplied, i.e. without the drive belt slipping with respect to the primary pulley. The greater this force reserve, the more the ratio which is actually effected, i.e. the FpFs ratio, can deviate from the KpKs ratio and the more quickly

the transmission ratio of the transmission can be changed. If the FpFs ratio can adopt a value of lower than 1, the safety factor Sf in the Master-Slave control should be at least equal to 1 divided by the lowest FpFs ratio which occurs, [FpFs]$_{MIN}$ for short.

The lowest FpFs ratio which occurs is generally lower than the lowest KpKs ratio which occurs, and consequently the control has to satisfy the following condition:

$$Sf \geq \frac{1}{[FpFs]_{MIN}} + C \tag{4}$$

in which C is a component in the safety factor Sf for the abovementioned compensation for the inaccuracy in the parameters.

If the control draws a distinction between equilibrium and non-equilibrium states of the transmission, at least in the equilibrium state it should satisfy the generally slightly less strict condition that:

$$Sf \geq \frac{1}{[KpKs]_{Low}} + C \tag{5}$$

The abovementioned aspects mean that in the known Master-Slave control, and more particularly in the case of the primary clamping force Kp in Low, a relatively high safety factor and therefore also relatively high clamping force levels have to be used. However, this control has the important advantage the torque which can be transmitted by the transmission is in principle determined exclusively by the secondary clamping force level, i.e. separately of the control of the transmission ratio with the aid of the primary clamping force level. As a result, the Master-Slave control can be set up in relatively simple form both in terms of software and in terms of hardware, yet can nevertheless quickly and accurately realize a desired change in the torque which can be transmitted or the transmission ratio.

In practice, the known transmission has in particular proven to be a reliable and efficient automatic transmission between

the engine and the driven wheels of a motor vehicle for
passenger transportation. In an application of this nature,
the efficiency of the drive as a whole and that of the
transmission in particular are considered as an essential if
5    not crucial characteristic of the vehicle. The efficiency of
the transmission is in this context inversely related to the
maximum level of the clamping forces. For example, a
frictional loss between drive belt and pulley increases with
said force level, as does wear to these components and in
10   particular the drive belt. Also, the power which is required
to generate a force, for example by hydraulic or electrical
means, generally increases with the level of this force.

It is an object of the present invention to improve the
efficiency of the transmission by reducing the level of the
15   higher of the two clamping forces which is required during
operation, in particular without at least significantly
detracting from the most important functional aspects and the
performance of the transmission.

According to the invention, an improvement of this nature is
20   realized in the design according to claim 1. The transmission
according to the invention is characterized in that, at least
when using a safety factor Sf of around 1.3, the KpKs ratio
in Low is at least equal to 1 but is no greater than the KpKs
ratio in Overdrive.

25   This higher KpKs ratio in Low compared to the known
transmission advantageously leads to a lower level of the
primary clamping force required. After all, the primary
running radius is lowest in Low, and therefore according to
equation (1) the required primary clamping force is then
30   highest, at least if the maximum torque supplied is identical
or lower in all the other transmission ratios, as is the case
when used in a motor vehicle. In other words, the KpKs ratio
in Low is the determining factor for in absolute terms the
highest level of the clamping force which is realized during
35   operation and on the basis of which the transmission needs to
be calculated. The lower this maximum force level which
occurs, the lower the demands which are imposed in mechanical
terms on the design of the transmission and the more cost-
effectively and efficiently the transmission can be produced

and operated.

It should be noted that all this is of particular relevance to the transmission provided with the Master-Slave control, since in that case the relatively high safety factor is used in equation (2). Moreover, at least in combination with the Master-Slave control and in equilibrium situations, the safety factor only has to be matched to the phenomena which it is to compensate for, such as the abovementioned inaccuracy in the determination of the clamping forces, which means that the abovementioned value of 1.3 is generally sufficient for this factor.

A KpKs ratio of 1 in Low in principle leads to a minimum and therefore optimum level of both the primary clamping force and the secondary clamping force. However, in particular in combination with the Master-Slave control, a KpKs ratio value in Low of greater than 1 may also be advantageous if the lowest value of the FpFs ratio which occurs during operation is lower than or at most equal to 1. The upper limit for the KpKs ratio in Low is in this context determined according to the invention in such a manner that it is at most equal to the KpKs ratio in Overdrive, so that the conventional shifting of the behavior of the transmission is retained. This means that in a non-equilibrium state in which a constant FpFs ratio is used, the speed with which the transmission ratio of the transmission changes remains accurately controllable and predictable, or at least that it does not have the tendency to adopt a possibly inadmissible value. It is preferable for the KpKs ratio in Overdrive itself to be at least slightly, for example at least 10%, higher than that in Low, so that a stable equilibrium state of the transmission is obtained, in which any variation in one of the clamping forces is of its own accord compensated for by a slight change in the transmission ratio.

According to the invention, the value of the KpKs ratio is certainly also important to the transmission efficiency in Overdrive. A reduction in this value according to the invention has a positive effect on the efficiency and robustness of the transmission, since these two aspects improve as the highest clamping force required, i.e. the

primary clamping force in Overdrive, decreases. For example, a frictional loss between drive belt and pulley decreases with a decreasing clamping force level, as does wear to these components. Also, the power which is required for generating
5   the clamping forces, for example by hydraulic or electrical means, generally also decreases with the force level that is to be generated. The efficiency of the transmission in Overdrive is therefore inversely related to the level of the primary clamping force. In this context, in particular the
10  value of the KpKs ratio in Overdrive is a decisive factor in the fuel consumption of a motor vehicle which represents the most important application for the transmission, on account of the fact that in such an application the transmission is generally in or close to Overdrive for a relatively long
15  time, if not most of the time. Therefore, the invention also relates to a transmission in which, at least when using a safety factor Sf of around 1.3, the KpKs ratio in Overdrive has a value in the range between 1.8 and the KpKs ratio in Low.
20  The transmission according to the invention advantageously makes use of transmission parameters on the equilibrium clamping force ratio, i.e. KpKs curve, of the transmission. It has been found from an analysis of this phenomenon, on which the invention is based, that in the particular case of
25  the transmission provided with a drive belt of what is known as the push belt type, the KpKs curve deviates unexpectedly and to a considerable extent from what could be expected in an initial, obvious approximation. More particularly, the analysis according to the invention reveals the influence on
30  said KpKs curve of the contact angle, i.e. the ratio between the tangent of the contact angle for the primary pulley and that for the secondary pulley and of the tangential coefficient of friction between push belt and pulley. In a more detailed refinement of the invention, the equilibrium
35  clamping force ratio, i.e. the KpKs curve, therefore has a numerical value in the range of 1.6 to 1.2 over the entire range of transmission ratios of the transmission, at least when using a safety factor Sf of around 1.3, and moreover preferably has a virtually linear profile. In a particularly

advantageous application of the invention, the KpKs ratio is virtually constant under the abovementioned conditions and has a value of approximately 1.3 in Low up to approximately 1.5 in Overdrive.

5   A transmission of this type in particular takes account of the influence of the absolute value of the contact angles on the functioning thereof, as will be explained in more detail below. A transmission of this type can achieve a considerable increase in efficiency without detracting, at least to a

10  significant degree, from the most important functional aspects and the performance of the transmission provided with the push belt. A linear profile of the KpKs curve is advantageous according to the invention since in that case the transmission, at every transmission ratio, will

15  advantageously react in more or less the same way to a change in the primary and/or secondary clamping force. This aspect is of benefit to the simplicity and robustness of the transmission control responsible for regulating the desired clamping forces.

20  The present invention provides a number of exemplary embodiments of the transmission, in which the said equilibrium clamping force ratio is realized in an advantageous way, which examples are described below with reference to the appended explanatory figures.

25

Figure 1 diagrammatically depicts a cross section through a continuously variable transmission provided with two pulleys and a drive belt in accordance with the prior art.
Figure 2 shows a simplified side view of the transmission

30  shown in Figure 1.
Figure 3 shows a cross section through the push belt which can preferably be used as the drive belt in the continuously variable transmission according to the invention.
Figure 4 shows a side view of a transverse element from the

35  push belt shown in Figure 3.
Figure 5 shows a detail of a pulley disk, and in particular its contact surface, as can be used in combination with the push belt shown in Figure 3 in the continuously variable transmission according to the invention.

Figure 6 illustrates the difference in clamping force between the primary pulley and the secondary pulley as a result of the transmission ratio.

Figure 7 uses a small part of a curved drive belt to
5   illustrate the relationship between a tensile stress therein and a force component which is exerted in the radially inward direction.

Figure 8 shows a diagram which plots the equilibrium clamping force ratio which has been theoretically approximated against
10  the transmission ratio of the known transmission with a constant contact angle of 11 degrees for both pulleys.

Figure 9 shows a diagram in which what is known as the contact angle contour for the primary and secondary pulleys is plotted against the transmission ratio, with the
15  theoretically approximated equilibrium clamping force ratio being equal to 1 irrespective of this transmission ratio.

Figure 10 illustrates the play of forces in the transmission provided with a drive belt of the push belt type and illustrates the difference in clamping force between the
20  primary pulley and the secondary pulley as a result of the transmission ratio.

Figure 11 illustrates, in a tangential cross section of the drive belt and a pulley, the play of forces in the contact between them under the influence of the axial clamping force
25  which is exerted.

Figure 12 shows a diagram in which the optimum contact angle contour, as it is known, is plotted for the primary pulley and the secondary pulley against the transmission ratio for a transmission provided with a drive belt of the push belt
30  type.

Figure 1 diagrammatically depicts a cross section through a continuously variable transmission 1 according to the prior art. The known transmission 1 comprises a primary pulley 2,
35  which can be driven by an engine (not shown) with a couple of forces Tp, and a secondary pulley 3, which can drive a load (not shown) with couple of forces Ts. Both pulleys 2 and 3 are provided with a pulley disc 21, 31, which is fixed to the respective pulley axle 20, 30, and with a pulley disc 22, 32

which can be displaced in the axial direction with respect to the said axle 20, 30. A drive belt 10, more particularly a push belt 10, is clamped between the pulley discs 21, 22, 31, 32, so that with the aid of friction mechanical power can be

5    transmitted between the two axles 20 and 30. An axially oriented force with which the drive belt 10 is clamped in place for each pulley 2, 3, referred to below as the primary clamping force Kp and the secondary clamping force Ks, respectively, is in this case realized by the application of

10   a hydraulic pressure in a respective pressure chamber 24, 34 of the two pulleys 2 and 3.
     The transmission ratio Rs/Rp of the transmission 1 is determined by the ratio between a secondary running radius Rs and a primary running radius Rp of the drive belt 10, i.e.

15   the effective radial position thereof between the pulley discs 21, 22, 31 and 32 of the respective pulleys 2 and 3. The said running radii Rp and Rs, and therefore the transmission ratio Rs/Rp, defined in accordance with the invention, of the transmission 1 can be varied by the

20   displaceable discs 22, 32 being moved in opposite axial directions over the respective pulley axles, 20, 30. In Figure 1, the transmission 1 is illustrated with a small transmission ratio Rs/Rp, i.e. with a relatively large primary running radius Rp and a relatively small secondary

25   running radius Rs.
     It should be noted that the transmission ratio Rs/Rp, the primary running radius Rp and the secondary running radius Rs are in a clearly defined and geometrically determined relationship with respect to one another, this relationship

30   being determined, inter alia, by the length of the drive belt 10, the distance between the axes of rotation of the respective pulleys 2, 3 and the maximum and minimum running radii Rp and Rs, so that these variables can be calculated from the others as desired.

35   Figure 2 shows a further, side view of the known transmission 1, with the primary pulley 2 with the primary axle 20 on the left-hand side of the figure and the secondary pulley 3 with the secondary axle 30 on the right-hand side of the figure. Unlike in Figure 1, in this figure the transmission 1 is now

illustrated with a relatively high transmission ratio Rs/Rp, in which the primary running radius Rp is smaller than the secondary running radius Rs, with the result that, during operation, the primary pulley 2 will have a lower rotational
5   speed than the secondary pulley 3. The drive belt 10 shown is what is known as a push belt 10, which comprises a virtually continuous series of transverse elements 11, only a few of which are shown, for the sake of simplicity, and at least one set 12 of a number of radially nested, continuous, flat and
10  thin metal rings.
    This push belt 10 is shown in more detail in Figures 3 and 4, Figure 3 showing a cross section through the push belt 10 and Figure 4 showing a side view of a transverse element 11 therefrom. The cross section shows a front view of the
15  transverse element 11, which is provided on either side with a recess, in each of which there is a set of rings 12. The set of rings 12 and the transverse element 11 retain one another in the radial or height direction, but the transverse elements 11 can move along the sets of rings 12 in the
20  circumferential direction thereof. Furthermore, the transverse elements 11 are provided with a protuberance in the circumferential direction of the push belt 10, also referred to as the projection 13, and with a recess 14 arranged in an opposite main side of the element 11, which
25  projection 13 and recess 14 serve to stabilize the series of transverse elements 11 in the push belt 10 with respect to one another.
    A bottom section 15 of the transverse element 11 tapers, so that adjacent transverse elements 11 can tilt with respect to
30  one another and the push belt 10 can describe an arc, such as where it is clamped between the pulley disks 21, 22, 31, 32 of the respective pulleys 2 and 3. It should be noted that the abovementioned effective radial position, i.e. the effective running radius Rp, Rs of the push belt 10,
35  substantially corresponds to a radial position of the topside of the bottom section 15 of the transverse element 11, which topside is also referred to as the tilting line 17 of the transverse elements 10, along which the latter are in contact with one another in the said arc. The bottom section 15 is

furthermore provided, on either side with what are known as
running surfaces 16, via which the transverse element 11 is
clamped between the pulley discs 21, 22; 31 32, the rotation
of a driving pulley 2 being transmitted via friction to the
5    clamped transverse elements 11. This may give rise to a
considerable pushing force between the transverse elements
11, with the result that they push one another forward over
the sets of rings 12 in the direction of the driven pulley 3.
Then, where the push belt 10 is clamped between the discs 31
10   and 32 of the driven pulley 3, the pushing force which is
present between the transverse elements 11 is virtually
completely transmitted via friction to the driven pulley 3.
Finally, the transverse elements 11 push one another back,
exerting a relatively low pushing force, from the driven
15   pulley 3 to the driving pulley 2. The sets of rings 12 in
this case ensure that the transverse elements 11 continue to
follow the path which is intended for the push belt 10.
     Figure 5 illustrates a detail of a pulley disk 43 on the
basis of a cross section through it as seen in the tangential
20   direction. A so-called contact surface 40 of the pulley disk
43, by means of which it comes into contact with a running
surface 16 of the transverse elements 11, is provided with a
curvature with an optionally variable radius of curvature
R40, with a contact angle $\lambda$, defined between a tangent line
25   41 in a point R on the contact surface 40 and the radial
direction 42, increasing as seen in the said radial
direction. Therefore, the contact surfaces 40 in the
transmission 1, as seen in the tangential cross section,
describe a contour which can be defined as the relationship
30   between the local contact angle $\lambda$ and the transmission ratio
Rs/Rp of the transmission 1. For each pulley 2, 3, the said
contour is referred to as the primary contact angle contour
$\lambda p(Rs/Rp)$ and the secondary contact angle contour $\lambda s(Rs/Rp)$,
respectively, with the fixed and moveable discs 21, 22, 31
35   and 32 of a pulley 2, 3 being provided with identical
contours. It is also preferable for the two pulleys 2 and 3
to be identical in form, i.e. to be provided with contact
angle contours $\lambda p(Rs/Rp)$ and $\lambda s(Rs/Rp)$ which are mirror-
symmetrical with respect to one another.

In order to be able to interact optimally with the curved
contact surfaces 40 of the pulleys 2 and 3, the running
surfaces 16 of the transverse element 11, as seen in the
cross section of the push belt 10 illustrated in Figure 3,

5    are provided with a curvature. In this case, a range of
contact angles λ which at least corresponds to contact angle
contours λp(Rs/Rp) and λs(Rs/Rp) defined by the contact
surfaces 40 of the pulleys 2 and 3, is defined in the contour
of the running surfaces 16.

10   The clamping force ratio, the KpKs ratio, which is required
for an equilibrium state of the transmission 1, i.e. for a
constant transmission ratio, originates from the equilibrium
condition whereby per pulley 2, 3 the tensile forces Ft
generated in the belt sets 12 of the drive belt 10 should be

15   equal to one another. This equilibrium condition is
illustrated in Figure 6. For each pulley 2 and 3, the tensile
force Ft is produced as a result of the radial forces Frp and
Frs, respectively, acting on the drive belt 10 in the radial
direction, which forces Frp and Frs are produced as a result

20   of the local contact angle λp, λs and the clamping force Kp,
Ks which is applied between the discs 21 and 22, 31 and 32
respectively, for each pulley 2, 3 and is substantially
axially oriented. When this is written out for the primary
pulley 2, the following relationship applies:

25

$$Frp = Kp * \tan(\lambda p) \qquad\qquad\qquad (6)$$

The radial forces Frp and Frs act on the running surfaces 16
of the transverse elements 11 over the length of the parts of

30   the drive belt 10 which are clamped between the pulley discs
21, 22, 31, 32 of the respective primary pulley 2 and
secondary pulley 3. The said length can be quantified for
each pulley 2, 3 as an angle which is enclosed by the clamped
part of the drive belt 10, designated here as the primary

35   belt angle αp and the secondary belt angle αs, respectively.
The radial forces Frp and Frs which are required for
equilibrium are then determined by summing, over the
respective belt angles αp and αs, the tensile force Ft per

unit of the belt angle $d\alpha$. When written out for the primary pulley 2, the following relationship therefore applies:

$$Frp = \int_0^{\alpha p} \frac{Ft}{Rp} * Rp * d\alpha \qquad (7)$$

5   The derivation of equation (3) is illustrated in Figure 7 on the basis of a small part of the belt set 12.

Equations (6) and (7) can be derived in a corresponding way for the secondary pulley 3, with the said equilibrium
10  condition for a constant transmission ratio whereby the tensile stress Ft which is generated applies to both pulleys 2 and 3, so that the following relationship applies to the equilibrium clamping force ratio KpKs:

$$15 \quad KpKs = \frac{Kp}{Ks} = \frac{\tan(\lambda s)}{\tan(\lambda p)} * \frac{\int_0^{\alpha p} Ft * d\alpha}{\int_0^{\alpha s} Ft * d\alpha} = \frac{\tan(\lambda s) * \alpha p}{\tan(\lambda p) * \alpha s} \qquad (8)$$

in which the belt angles $\alpha p$ and $\alpha s$ vary as a function of the respective running radius Rp, Rs and therefore also as a function of the transmission ratio Rs/Rp. A relationship of
20  this nature between the belt angles $\alpha p$ and $\alpha s$ and the running radii Rs and Rp is determined by the geometry of the transmission 1 and may, for example, be approximated relatively accurately as:

$$25 \quad \alpha_p = \pi + 2 * \arcsin\left(\frac{Rp - Rs}{2 * Rp_{MAX}}\right) \text{ and } \alpha s = 2\pi - \alpha p \qquad (9)$$

with Rs(Rp, Rp$_{MIN}$, Rp$_{MAX}$):

$$2 * \sqrt{(2 * Rp_{MAX})^2 - (Rs - Rp)^2} + \pi * (Rs + Rp) +$$

$$2 * \arcsin\left(\frac{Rs - Rp}{2 * Rp_{MAX}}\right) * (Rs - Rp) = 2 * \sqrt{(2 * Rp_{MAX})^2 - (Rp_{MAX} - Rp_{,MIN})^2} +$$

$$\pi * (Rp_{MAX} + Rp_{min}) + 2 * \arcsin\left(\frac{Rp_{MAX} - Rp_{MIN}}{2 * Rp_{MAX}}\right) * (Rp_{MAX} - Rp_{MIN})$$

$$(10)$$

where $Rp_{MIN}$ is the smallest primary running radius Rp which occurs and $Rp_{MAX}$ is the largest primary running radius Rp which occurs. In the derivation of equations (9) and (10), it has been assumed that the two pulleys 2 and 3 are positioned as close together as possible in the radial direction, as is the case, for example, in Figure 6, but as is also generally desired in motor vehicles.

The solution, which is to be determined iteratively or numerically, to equations (8), (9) and (10) for the KpKs ratio in relation to the transmission ratio Rs/Rp, with the contact angles $\lambda p$ and $\lambda s$ having a constant and equal value – in this example 11 degrees – is given in Figure 8.

It can be concluded from the above analysis, which, incidentally, is independent of the type of drive belt 10, meaning that it applies not only to the push belt 10 shown in Figures 2-4 but also to a rubber V belt, a metal chain or the like, that the KpKs ratio can be influenced by selecting values for the primary contact angle $\lambda p$ and/or the secondary contact angle $\lambda s$ which differ from one another. The ratio between the contact angles $\lambda p$, $\lambda s$ as a function of the transmission ratio Rs/Rp of the transmission, with the equilibrium ratio of the clamping forces Kp and Ks – referred to as the KpKs curve – advantageously being equal to 1 for all the transmission ratios Rs/Rp, should in this case satisfy equation (4) with Kp/Ks=1:

$$\frac{\tan(\lambda s)}{\tan(\lambda p)} = \frac{\alpha s}{\alpha p} \qquad (11)$$

Incidentally, it follows from equation (11) that the contact angles $\lambda p$, $\lambda s$ should be equal in value in the transmission ratio Rs/Rp in which the belt angles $\alpha p$ and $\alpha s$ – and therefore also the running radii Rp and Rs – are equal to one another.

One possible solution to equation (11) is shown in the diagram in Figure 9, in which the respective contact angle $\lambda p$, $\lambda s$ for the primary pulley 2 and the secondary pulley 3 is plotted against the transmission ratio Rs/Rp in the so-called contact angle contours $\lambda p$(Rs/Rp) and $\lambda s$(Rs/Rp). The

theoretically approximated KpKs ratio is in this case, therefore equal to 1 in all possible transmission ratios Rs/Rp. The diagram shown in Figure 9 applies to a typical transmission 1 with a smallest primary running radius $Rp_{MIN}$ of
5   approximately 30 mm and with a largest primary running radius $Rp_{MAX}$ of approximately 75 mm combined with an equal and smallest possible radial dimension of the pulleys 2 and 3. Although the analysis discussed above does suggest this, the abovementioned and apparently optimum contact angle contours
10  $\lambda p(Rs/Rp)$ and $\lambda s(Rs/Rp)$ do not in all cases form the most ideal solution for an improved efficiency of the transmission 1 via influencing the equilibrium clamping force ratio KpKs, at least not for the present transmission 1 provided with the push belt 10.
15  Furthermore, in particular in combination with the Master-Slave control, it may be highly advantageous to select the KpKs ratio to be greater than 1, as has already been described above. Secondly, the Applicant has discovered, by means of an analysis of this phenomenon on which the
20  invention is based, that in the particular case of the transmission 1 provided with a push belt 10, during operation of this transmission, a unique play of forces is produced in the push belt 10, which is of considerable influence on the equilibrium ratio of the clamping forces Kp and Ks. Moreover,
25  in this respect it has been found that in the battle for an improved efficiency of the transmission 1 by influencing equilibrium clamping force ratio KpKs as result of the contact angles $\lambda p$ and $\lambda s$ being adapted, the mechanical load on the push belt 10, and in particular a fatigue load of the
30  sets of rings 12 thereof, but also, for example, the dynamic performance of the transmission 1 is influenced.
In view of the complexity of these phenomena and the interaction between them, an analytical description thereof cannot be realized or can only be realized with extreme
35  difficulty. However, according to the invention a qualitative analysis is sufficient, and an improved transmission design is proposed on the basis thereof.
The qualitative analysis according to the invention is based on a pushing force Fd between the transverse elements 11, a

tensile force Ft in the rings and a normal force in the radial direction Fr between the individual transverse elements 11 and the sets of rings 12 as occurs on the pulleys 2 and 3. In this context, the pushing force Fd is

5 substantially responsible for the transmission of torque between the pulleys 2 and 3, with the sets of rings 12 and the transverse element 11 being held together and forced into a desired path together with the pulleys 2 and 3. According to the invention, the following equation (written out here

10 for the primary pulley) applies to each pulley 2, 3:

$$Kp * \tan(\lambda p) = \int \frac{\delta[Ft(\alpha p)] - \delta[Fd(\alpha p)]}{Rp} * Rp * \delta\alpha p \qquad (12)$$

or, for equilibrium between the two pulleys 2 and 3:

15

$$KpKs = \frac{\tan(\lambda s)}{\tan(\lambda p)} * \frac{\int \delta[Ft(\alpha p)] - \delta[Fd(\alpha p)] * \delta\alpha p}{\int \delta[Ft(\alpha s)] - \delta[Fd(\alpha s)] * \delta\alpha s} \qquad (13)$$

It follows from these equations that in the case of the push belt 10, the equilibrium clamping force ratio KpKs is

20 influenced not only by the contact angles $\lambda p$, $\lambda s$ but also by the pushing force Fd and more particularly by the distribution thereof over the belt angle $\alpha p$, $\alpha s$. According to the invention, it is stated on the basis of equation (11) that the pushing force Fd partially compensates for the

25 radial forces Frp and Frs generated by the respective clamping force Kp, Ks. Moreover, the transverse elements 11 and the sets of rings 12 are in frictional contact with one another, so that as a result of a difference in speed between them a build-up or decrease in tensile force Ft occurs in the

30 rings, which phenomenon, in the case of the push belt 10, further complicates an analytical solution to the equilibrium clamping force ratio KpKs. The present invention therefore proposes a qualitative analysis.

The qualitative effect is illustrated in Figure 10 and is

35 based on the knowledge that during operation of the transmission 1, when a torque Tp is being transmitted between

the pulleys 2 and 3, for each pulley 2, 3 the belt angle $\alpha_p$, $\alpha_s$ is composed of two successive parts. In a first part of the belt angle, known as the creep angle K, the pushing force Fd between the transverse elements 11 of the push belt 10 is
5   built up or reduced in the frictional contact with the pulley 2, 3. In a second part of the belt angle, known as the rest angle R, the pushing force Fd is approximately constant and at least virtually equal to zero on the driving pulley – in this case the primary pulley 2 – and equal to a maximum value
10  Fd-max on the driven pulley – in this case the secondary pulley 3. The creep angle K over which the build-up or reduction takes place under identical conditions – i.e. an identical coefficient of friction and normal force between the transverse element 11 and the respective pulley 2, 3 – is
15  of approximately corresponding length, so that at a relatively high clamping force the creep angle K is relatively small with respect to the total belt angle $\alpha_p$, $\alpha_s$. All this is reproduced in Figure 10 for the lowest transmission ratio, i.e. Overdrive, with the double arrows
20  indicating the level of the pushing force Fd which prevails locally between adjacent transverse elements 11 in the push belt 10, but not the direction of the pushing force Fd, since this is oriented in the longitudinal direction of the push belt 10. The figure also shows a possible curve for the
25  tensile force Ft in the set of rings over the circumference of the push belt 10, which in this example counteracts the transfer of the torque Tp. After all, the tensile force has the highest Ft-max on the side of the push belt 10 where the highest pushing force Fd-max is present and the lowest level
30  Ft-min on the side of the push belt 10 where the lowest pushing force Fd, specifically virtually no pushing force, is present. The curve of the tensile force Ft may, incidentally, also be precisely the opposite, depending in particular on the level of the torque Tp.
35  In qualitative terms, it follows from Figure 10 that, at least in Overdrive, the equilibrium clamping force ratio KpKs, in the case of the push belt, has a value which is significantly different than what could be expected according to equation (8). After all, a cumulative pushing force Fd

over the primary belt angle $\alpha_p$ is significantly lower than the cumulative pushing force Fd over the secondary belt angle $\alpha_s$. The result of this is that the optimum contact angle contours $\lambda_p(Rs/Rp)$ and $\lambda_s(Rs/Rp)$ derived above do not apply

5 to the transmission 1 provided with the push belt 10.

Furthermore, it follows from the above analysis that the equilibrium clamping force ratio KpKs can not only be influenced by the ratio between the primary and secondary contact angles $\lambda_p$ and $\lambda_s$ but also can be reduced in Overdrive

10 by shortening the rest angle R or lengthening the creep angle K at least on one of the two pulleys 2, 3. For the secondary pulley 3, this will lead to a decrease in the cumulative pushing force Fd over the secondary belt angle $\alpha_s$, while for the primary pulley 2 it will lead to an increase in this

15 cumulative pushing force, which in both cases according to equation (12) leads to a reduction in the equilibrium clamping force ratio KpKs. In Overdrive, a shortened rest angle R on the secondary pulley 3 is not the most advantageous option, since in this case this is determined by

20 the desired safety factor Sf, which in the context of the present invention is considered as a given boundary condition. On the other hand, according to the invention it is advantageously possible to have a beneficial influence on the KpKs by lengthening the creep angle R on the primary

25 pulley 2, at least for the primary running radius Rp in Overdrive.

According to the invention, the creep angle can be lengthened by at least surprisingly making the transmission of force between the primary pulley 2 and the push belt 10 less

30 efficient, for example by selecting a lower effective coefficient of friction $\mu$ between them (cf. also equation (1)). According to accepted theory, the coefficient of friction $\mu$ of a lubricated frictional contact as is customarily used in the present transmission can be reduced

35 by the design of the pulley 2, for example by lowering the contact pressure between push belt 10 and pulley 2 - for example by using a relatively large radius of curvature R40 for the primary pulley disks - or by lowering the surface roughness of the pulley disks 21 and 22.

The abovementioned measures relate in particular to the contact point between the push belt 10 and the primary pulley 2 at a transmission ratio Rs/Rp of less than 1, i.e. to a relatively large primary running radius Rp, more particularly the largest primary running radius Rp which determines the transmission ratio Overdrive. At the other positions in the transmission 1, i.e. at a relatively small primary running radius Rp, such as for example the primary running radius Rp in Low, and at any desired running radius Rs on the secondary pulley 3, by contrast, it is advantageous not to use the abovementioned measures and to make the transmission of force between push belt 10 and pulley 2, 3 as efficient as possible. On the one hand since at the abovementioned relatively small primary running radius Rp and an associated relatively large secondary running radium Rs, i.e. in a transmission ratio Rs/Rp close or equal to the transmission ratio Low, the equilibrium clamping force ratio KpKs is still virtually equal to 1, so that the increase in efficiency will be relatively low, and on the other hand since the effectiveness of the transmission of force is crucial to the maximum torque which can be transmitted at a given clamping force Kp or Ks not only at the abovementioned relatively small primary running radius Rp but also at a relatively small secondary running radius Rs.

The measures discussed above may be employed independently of or in addition to the application of the contact angle contours λp(Rs/Rp) and λs(Rs/Rp). According to the invention, however, a more detailed analysis reveals that the KpKs curve which in principle is the most optimum with regard to the transmission efficiency, with the equilibrium ratio of the clamping forces constantly equal to 1, requires contact angle contours λp(Rs/Rp) and λs(Rs/Rp) which cause other functional aspects of the transmission 1 to deteriorate. More particularly, the total required difference between the smallest and largest contact angles λp, λs may become disadvantageously large. On the one hand, the smallest contact angle λp, λs required may, for example, be so small that the radial component Fr of the clamping force Kp, Ks is too small to overcome friction in the radial direction

between the push belt 10 and the pulley 2, 3, as a result of which it would be impossible to change the transmission ratio of the transmission 1. On the other hand, the largest contact angle $\lambda p$, $\lambda s$ required may, for example, be so large that the

5   sets of rings 12 are excessively loaded by the radial component Fr of the clamping force Kp, Ks. A description of such phenomena can be found, for example, in EP-0 798 492 and the Dutch patent application 1022157 in the name of the Applicant, which was not published before the priority date

10  of the present application.
    Yet another disadvantageous consequence of a large difference between the smallest and largest contact angles $\lambda p$, $\lambda s$ required is that the contact surfaces 40 of the pulleys 2 and 3 and the running surfaces 16 of the transverse elements 11

15  have to be relatively strongly curved in order to allow a difference of this nature to be realized within a limited dimension. As a result, the contact stress between them may adopt an undesirable or even inadmissible value. As has already been discussed above, the alignment of the

20  transmission 1 is also determined by the contact angle contours $\lambda p(Rs/Rp)$ and $\lambda s(Rs/Rp)$ and is a limiting factor therefor.
    In the empirical approximation according to the invention which incorporates these aspects, the contact angles, at

25  least in Overdrive, have to satisfy the following condition:

$$1 < \frac{\tan(\lambda p)}{\tan(\lambda s)} \leq 1.6 \qquad (14)$$

More particularly, this ratio moreover in Low satisfies the

30  following condition:

$$0.6 \leq \frac{\tan(\lambda p)}{\tan(\lambda s)} < 1 \qquad (15)$$

The analysis discussed above merely states a condition for

35  the ratio between the contact angles $\lambda p$ and $\lambda s$ but does not yet give the optimum values for these parameters. According to the invention, these optimum values are found as follows.

According to the invention, the lower limit for the contact angles $\lambda p$ and $\lambda s$, on the one hand, is preferably selected to be as low as possible, since as a result the radial force Frp, Frs and therefore also the tensile force Ft in the sets
5  of rings 12 will advantageously be low. After all, said tensile force Ft makes no contribution or scarcely any contribution to the transmission of the torque Tp supplied, while the sets of rings 12 are subjected to mechanical load by the tensile force Ft. On the other hand, according to the
10 invention, under all circumstances it must be possible for the drive belt 10 to be displaced in the radial direction between the pulley disks 21, 22, 31, 32, in order to allow the transmission ratio Rs/Rp to be altered. For this purpose, said radial force Frp must be able to overcome at least a
15 friction Fw between the drive belt 10 and a pulley 2, 3. Written out for the primary pulley 2, the following relationship then applies:

$$\frac{Kp}{\cos(\lambda p)} * \tan(\lambda p) = \frac{Frp}{\cos(\lambda p)} = Fw > Fw = \mu_R * Fn = \mu_R * \frac{Kp}{\cos(\lambda p)} \qquad (16)$$

20
where $\mu_R$ is a coefficient of friction which is measured in the radial direction in the contact between a running surface 16 of the drive belt 10 and the contact surface 40 of a pulley disk 43, and where Fn is normal force in that contact.
25 Equation (16) is schematically illustrated in Figure 11, which shows the forces – Kp, Fw, Frp, Fn – which are active within said contact. Equation (16) gives rise to the condition that a contact angle $\lambda$ must be greater than the arc-tangent of the radial coefficient of friction $\mu_R$. In the
30 lubricated metal/metal contact between the pulleys 2 and 3 and the drive belt 10 of the transmission, a maximum value of approximately 0.12 typically applies for $\mu_R$. Therefore, according to the invention the primary contact angle $\lambda p$ in Low and the secondary contact angle $\lambda s$ in Overdrive is
35 preferably approximately equal to 7 degrees. The complete contact angle contours $\lambda p(Rs/Rp)$ and $\lambda s(Rs/Rp)$ can then be approximated iteratively using equations (13).
Yet another suitable boundary condition for solving the

boundary condition required by equation (13) may be that the
contact angle contours $\lambda_p(R_s/R_p)$ and $\lambda_s(R_s/R_p)$ for each
pulley 2, 3 is a continuous curve, respectively rising
continuously on the secondary pulley 3 and falling
5   continuously on the primary pulley 2. Finally, it may be
advantageous if the disks 21, 22, 31 and 32 of the pulleys
are shaped identically, which is advantageous in particular
in terms of production and assembly engineering
considerations.
10  With the contact angle contours $\lambda_p(R_s/R_p)$ and $\lambda_s(R_s/R_p)$ in
which the abovementioned aspects and factors, such as the
largest and smallest permissible values for the contact
angle, the lowest permissible value for the radius of
curvature R40 of the contact surfaces 40 and the radius of
15  curvature R16 of the running surfaces 16, as well as the
alignment of the transmission, are taken into account during
the determination of the optimum KpKs curve, the latter has a
more or less linear curve from 1.5 in Overdrive to 1.3 in Low
when using a safety factor of approximately 1.3. The diagram
20  shown in Figure 12 plots both this optimum KpKs curve and the
associated optimum contact angle contours $\lambda_p(R_s/R_p)$ and
$\lambda_s(R_s/R_p)$ against the transmission ratio $R_s/R_p$. The diagram
has been approximated by an empirical route, taking account
of all the above-discussed characteristics and properties of
25  the transmission 1 provided with a drive belt 10 of the push
belt type. It can be seen from this figure that the smallest
contact angles both for the primary pulley $\lambda_p$ and for the
secondary pulley $\lambda_s$ are both just slightly larger than
7 degrees, specifically approximately 7.3 degrees, and that
30  the largest primary contact angle $\lambda_p$ is approximately
10 degrees and the largest secondary contact angle $\lambda_s$ is
approximately 9 degrees. The optimum ratio of the tangent of
the contact angles $\lambda_p$, $\lambda_s$ is therefore approximately 1.4 in
Overdrive and approximately 0.8 in Low.